rod foot is about 70 per cent of the width of the bearing, and its thickness J to f in. greater than the diameter of the bolts. The stress on the bolts at the bottom of the thread may be 4500 to 5000 Ib. per square inch.

Piston-rods.—These are made of mild steel. The stress in the body of the rod should be 2800 to 3000 Ib. per square inch, and the stress at the bottom of the thread for the piston nut should be 5000 to 5500 Ib. per square inch. The part which enters the piston should be partly parallel and partly taper, the parallel part having the same diameter as the top of the threads of the screwed part, which usually has four threads per inch. The taper is usually one in four on the diameter, and the larger end of the taper is a little less than the diameter of the body of the rod, leaving a shoulder at that point. The shoulder does not, however, butt against the under side of the piston, a slight clearance of about *-5%* to xV in. being left between these two parts.. The attachment to the crosshead is frequently precisely similar in design to that at the piston end.

Thrust Block.—Recently a new type of thrust bearing (illustrated in Vol. IV, p. 9), invented by Mr. Anthony G. M. Michell, has come into successful use. There is only one collar, and the length and weight of the whole fitting is much reduced. Its design depends upon the application of principles first laid down by the late Professor Osborne Reynolds as a result of his investigations into the theory of lubrication.

Briefly put, it means that considerable loads per unit of area can be sustained if the two rubbing surfaces are not parallel, but have a slight inclination towards each other in the direction of motion, so that the film of oil between the surfaces is wedge-shaped, the thicker end being at the entrance edge. The two surfaces together form a kind of pump which continually renews the film of oil between them, and by maintaining the pressure of the oil prevents, even under very high loads, metallic contact. Experience showed that one of these surfaces must be free to take up the exact inclination required by the varying conditions of load, surface speed, and viscosity of oil. With these conditions the coefficient of friction is

reduced to about 0-0018, or *•%•$* of the value that would exist under similar conditions of loading if the surfaces were maintained parallel to each other.

Crank-shafts.—In practice these shafts are never designed *ab initio.* For many years certain rules laid down by the Board of Trade, and such societies as Lloyd's Register, the British Corporation for Survey and Register of Shipping, and the Bureau V6ritas International Register of Shipping, were worked to by designers.

Recently a committee was formed, entitled British Marine Engineering Design and Construction Committee, composed of gentlemen interested in the design and manufacture of marine engines, and of representatives of the societies above mentioned. The object of this committee was to standardize, as far as possible, design and construction, and new rules governing the sizes of shafts, having been agreed upon, were issued.

So far as reciprocating engines are concerned, the rules are as follows: